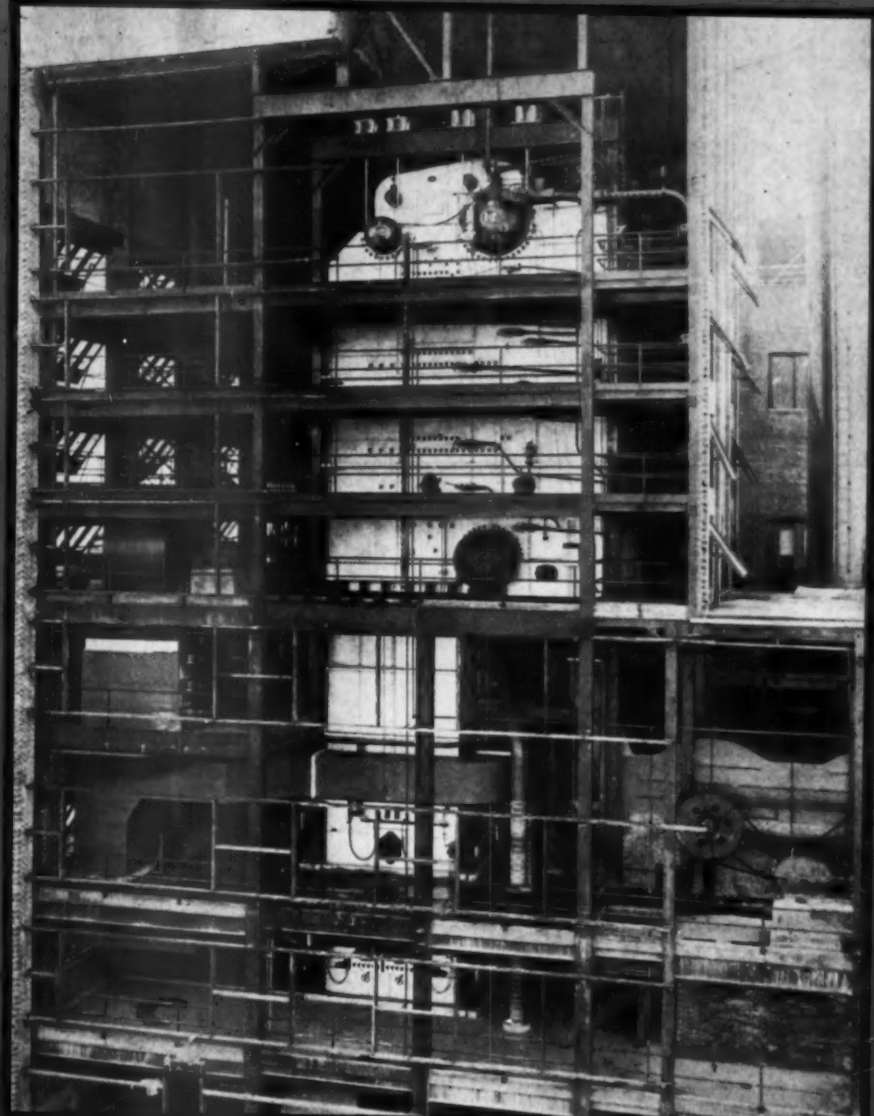
COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

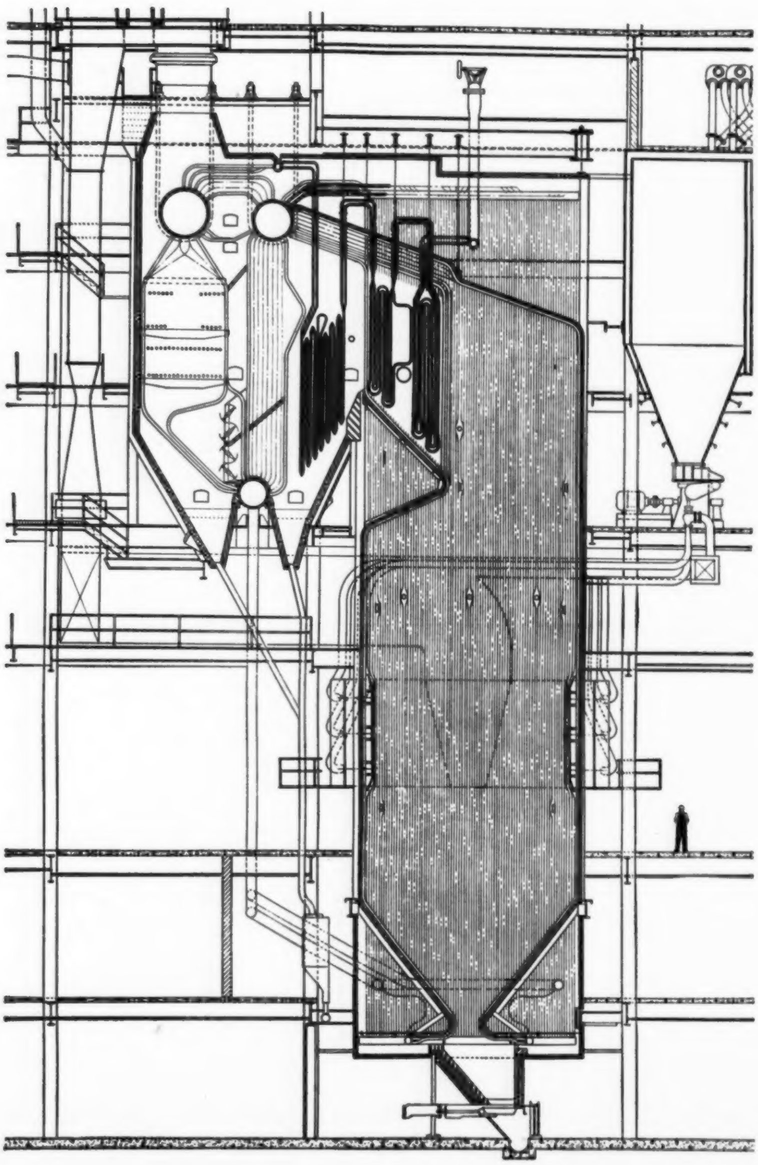
March, 1948



It is seldom possible to photograph a complete large modern steam generating unit. This one in a southern paper mill burns black liquor for chemical recovery and generates by-product process steam

Safety Valves for Stationary Power Boilers ▶

Factors in Selection of Steam Generating Units ▶



COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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FOR MARCH 1948

FEATURE ARTICLES

EDITORIALS

DEPARTMENTS

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EDITORIAL

Omission of I. D. Fans

Serious thought is being given by some power station designers to omission of the induced-draft fans in order to cut down on maintenance, reduce auxiliary power consumption and save on initial expense. While in no cases to date have these fans actually been left out, provision has been made in at least one new high-pressure station now under construction for trial operation without the induced-draft fans in service.

Obviously, with the induced-draft fans omitted, substantial pressure must be carried in the furnace and this requires special casing design to prevent leakage at ash-hopper connections, soot-blower wall boxes and observation doors—in other words to insure absolute tightness. This means some increased initial cost and perhaps some operating inconvenience. In some marine installations furnaces have been operated under pressure by employing double casings. Power required to drive the forced-draft fan would be stepped up although not enough to offset that which would be required by an induced-draft fan.

All these factors must be weighed in the individual case to determine whether the net saving is worthwhile; and the proposed operation without induced-draft fans in the above-mentioned case will be closely followed when that plant goes into service some two years hence.

The Annual Fuel Rate per Kilowatt-Hour

There has been practically no change in the average annual coal rate per kilowatt-hour output of utility plants, for the country as a whole, during the last five years. This is in contrast to the steadily decreasing rates from 1920 to 1942, of 3 pounds to 1.3 pounds. Of course, many individual stations have surpassed this latter figure of average performance.

Several factors contributed to this practically stationary fuel rate. In such new power construction as was permitted during the war, capacity and speed of installation took precedence over performance; war production often did not permit taking units out of service for usual overhaul; and, in general, the quality of available coal had deteriorated. For a considerable period following the war, the urgency of providing additional capacity to meet accelerated peacetime production, in many cases, precluded taking advantage of certain more recent technological advances in design.

However, with new capacity now overtaking the rate of load increase, attention is being given to improved station efficiency. In many installations now being laid down, or under construction, the answer seems to lie in a combination of higher steam temperatures and reheat, with the latest boilers and turbines especially designed to meet this combination most effectively; also designed to handle a wide range in coal quality. The number and total capacity of these new installations will not be sufficient to affect greatly the average fuel rate for the whole country for some time to come, but by 1950 their influence should begin to be felt, although improvement from now on will not be as marked as in the past, due to application of the so-called law of diminishing returns.

Coal Reserves

Of late, doubt has been expressed, in informed circles, as to the correctness of the much-cited Geological Survey estimate that there are sufficient coal reserves in the United States to last some three thousand years. It has been pointed out that this estimate, based on the measured extent and thicknesses of deposits, does not take into account mining losses; hence it does not represent the recoverable tonnage, which is probably about two-thirds that existing in the seams. Furthermore, the rate of energy consumption with all types of fuel is steadily increasing; and with natural oil supplies limited, future synthetic production of oil from coal will make further substantial inroads on the coal reserves. That is, although present commercial efforts are being directed largely to the synthetic production of oil from natural gas, the supply of surplus gas is limited and the eventual major source must be coal, extraction from which is still in the pilot plant stage.

Taking all these factors into consideration, revised estimates of the length of time we can depend on coal vary all the way from two hundred fifty to seventeen hundred years. This is indeed a wide spread, with the true figure probably somewhere between these two extremes. However, those now living need be little concerned as to the actual number of years that coal will last, for long before the reserves are exhausted nuclear energy will probably have become a major source for meeting many of our energy requirements. For the present, it is reassuring to know that the United States possesses about half the world's known coal resources; that these will last many years, even at a greatly accelerated rate of consumption; and that active steps are now being taken to make coal, as well as gas, a source of synthetic oil and gasoline supply to augment that obtainable from our limited natural sources.

Safety Valves for Stationary Power Boilers

Safety valves are a subject about which much could be written, but in the following the author will attempt merely to clarify some of the points concerning which confusion sometimes exists in the minds of operating men, and to explain certain requirements of the A.S.M.E. Code for Power Boilers.

By JOHN R. KRUSE

Combustion Engineering Co., Inc.

IN GENERAL, a safety valve is designed so that it will open automatically at a predetermined pressure and relieve any excess pressure in the boiler or superheater, or both. There are several classifications as to types of safety valves among which are the following:

1. A spring-loaded safety valve is one in which the disk is held against the valve seat by means of a compression coil spring. The amount of compression of the spring determines the pressure at which the valve is set to open and relieve the excess pressure.
2. A lever-weighted type of safety valve is one in which the disk is held against the seat by the action of a weight hung on a lever pivoted on a fulcrum. This type of safety valve is not permitted on power boilers by the A.S.M.E. Boiler Code.
3. A pilot-activated safety valve is one in which the disk is held against the seat by steam pressure and controlled in operation by a pilot-actuator valve. Usually the pilot valve is on the boiler drum and actuates the working safety valve mounted on the superheater.
4. A power-control safety valve is one in which the disk is actuated by some mechanical or electrical means, and functions in relation to steam pressure.
5. A duplex safety valve has twin disks, seats and springs mounted within a single body.
6. An extra "working" safety valve is a somewhat recent term applied to a safety valve mounted on the superheater outlet, or the boiler drum, and having a stop valve between the safety valve and the source of the steam. The relieving capacity of such a valve is in addition to that specified by the Boiler Code. By having a stop valve ahead of the safety valve a means is provided for removing it for repairs or replacement without shutting down the boiler.

Requirements of Safety Valves

There are certain requirements as to the number, sizes, capacities, setting pressures, etc., of safety valves as prescribed by the Power Boiler Code of the A.S.M.E., which will here be referred to merely as "The Code." Among these requirements are the following:

All power boilers having 500 sq ft, or more, of heating surface are required to have at least two safety valves on the boiler drum and at least one safety valve on the superheater, when a superheater is included in the steam generating unit.

CAPACITY—As to relieving capacity in pounds of steam per hour, there are two *minimum* values to be met, the greater of which governs, according to the Code.

One such minimum is that sufficient safety valve capacity has to be provided to relieve all the steam which the unit can generate when operating at maximum capacity. This maximum is equivalent to the combined total of all the safety valves on the drum, or drums, and on the superheater, within the limitations set up by the Code. This total capacity is also based on the requirement that the pressure in the boiler will not increase to more than 6 per cent above that of the highest set valve, and in no case more than 6 per cent above the maximum allowable pressure of the boiler. This maximum rated capacity is that given by the boiler manufacturer.

The other minimum capacity required is based on certain specified values proportional to the total steam generating surface of the boiler plus another value proportional to the total heating surface of the water walls. Both of these values depend on the method of firing. This minimum capacity can be determined by the expression

$$E = H_1C_1 - H_2C_2$$

in which

E = total required relieving capacity of all the safety valves, in pounds per hour, which is 90 per cent of the actual relieving capacity at the specified setting pressure as determined by the valve manufacturer in accordance with Code stipulations.

H_1 = total heating surface of the boiler, including that of the tubes, drums and headers, exposed to the products of combustion.

H_2 = total heating surface of water wall tubes and headers, exposed to the products of combustion.

The values of C_1 and C_2 vary with the general class of boiler and the method of firing, and are as follows:

	Water-tube Boilers			Fire-tube Boilers		
	Hand Fired or Waste Heat	Stoker Fired	Oil, Gas or Pulverized Coal Fired	Hand Fired or Waste Heat	Stoker Fired	Oil, Gas or Pulverized Coal Fired
C_1	6	8	10	5	7	8
C_2	8	12	16	8	10	14

Safety Valve Mountings

Safety valves can be mounted singly on separate nozzles, or single valves can be mounted in combinations of

two valves on a Y-base on a single nozzle. When only two valves are mounted singly on a boiler drum, the capacity of the smaller valve must be at least one-half that of the second valve. When two valves are mounted on a Y-base, or in the case of a duplex valve, both must be of the same capacity; although they may be set at slightly different pressures.

No steam outlet connection, other than for a safety valve, is permitted on any required safety valve nozzle, except for a soot blower, which can be attached to the safety valve connection of the superheater only.

The maximum distance between the inlet flange of a safety valve and the flange of the nozzle to which it is attached cannot be more than the face-to-face dimension of a standard T-fitting (A.S.A. Standard) for the required pressure. Moreover, all safety valves must be so attached as to have their stems in the vertical position.

Locations

When there is no superheater, all of the required safety valves must be connected directly to the steam drum or shell. However, when the steam generating unit includes an integral superheater, without any intervening stop valves, at least 75 per cent of the total required relieving capacity must be that of the safety valves mounted on the steam drum or shell. While the capacity of the safety valve mounted on the superheater can be greater than 25 per cent of the total required relieving capacity, the Code allows it to be credited with only a maximum of 25 per cent.

The Code requires at least one safety valve on all superheaters, but does not specify any minimum capacity of such valve or valves. However, for adequate protection of the superheater under all conditions, it is common practice to provide a minimum of 20 per cent of the total steam the unit will generate, as the relieving capacity of the superheater safety valves.

Setting Pressures

Although the Code has certain requirements as to the setting pressures for safety valves, in relation to design pressure and range of settings, practice in this respect goes far beyond the Code requirements. It is, therefore, felt that this subject can be best covered, not alone as to Code requirements, but also as a discussion of general practice.

The Code requires that the highest set safety valve on a boiler shall be at a pressure not greater than three per cent above the maximum allowable (design) pressure of the entire steam generating unit. It also specifies that the range between the lowest set valve and the highest set valve shall be not more than 10 per cent of the pressure of the highest set safety valve; also that the setting pressure of the lowest set valve shall not be greater than the maximum allowable pressure of the boiler.

The Code does not require that the safety valve on the superheater shall be set at a lower pressure than the lowest set valve on the boiler drum. It is common practice, however, always to set the superheater safety valves to relieve at a pressure sufficiently lower than that of those on the drum, so as to assure adequate protection of the superheater under all conditions. This differential setting between valves on drums and those on superheaters varies from about two or three pounds on small, low-pressure boilers to as much as 100 pounds or more on large high-pressure, high-temperature steam generating units. Much depends upon the amount of pressure drop through the superheater; also on the steam requirements and the actual blowdown pressure percentage of the safety valves. There is a wide difference of opinion among operating engineers as to the proper relation between the setting of safety valves on the superheater and on the drum.

Where possible there should be sufficient difference between the desired operating pressure and the set pressure of the lowest set safety valve. This will vary from about $3^1/_2$ to 5 per cent of the set pressure.

The per cent of blowdown also has to be taken into consideration in the setting of the safety valves. This has definite bearing on the actual design pressure of the steam generating unit.

For complete information concerning the sizes, settings, etc., of safety valves for both the steam drums and superheaters, one should consult the A.S.M.E. Code for Power Boilers, but the following typical examples may serve toward a clearer understanding of the subject.

Examples

No. 1—Assume a boiler without superheater, with 250 psi maximum design pressure and requiring one 2-in. and two 4-in. safety valves. The maximum operating pressure is desired.

Since 3 per cent of 250 = 7.5 psi, for good safety valve operation, the maximum operating pressure would be about 250 − 7, or 243 psi. The highest set pressure of one safety valve must be 250 psi. The other two can be set up to 3 per cent above 250 psi or about 257 psi. However, the probable setting of all three safety valves would be.

One 2-in. set at 250 psi, closing at 241 psi
One 4-in. set at 252 psi, closing at 243 psi
One 4-in. set at 255 psi, closing at 246 psi

These values are based on blowdown at $3^1/_2$ per cent.

From the above it will be seen that the pressure will reduce 2 lb under the 243-psi normal operating pressure, each time the safety valve pops.

If the same boiler had a superheater and the maximum capacity was the same, the 2-in. safety valve would likely be set on the superheater outlet. The same setting pressures could be used since the 2-lb differential between the superheater safety valve and the lowest set-down valve would probably be sufficient to assure steam flow through the superheater under all conditions.

No. 2—Consider a boiler and superheater of 900 psi design pressure, with a pressure drop through the superheater of 35 lb at maximum rating, the required safety valves being one 3-in. on the superheater and two 3-in. on the steam drum.

The maximum recommended operating pressure would be 3 per cent under 900 psi, or 873 psi at the boiler drum and 873 − 35 = 838 psi at the superheater outlet at maximum rating.

To meet Code requirements the lowest set valve on the drum would have to be set at 900 psi. The second could be set as high as 927 psi. The highest recommended setting for the superheater safety valve would be about 10 lb under the lowest set valve on the drum, or 890 psi. This 10-lb differential is not a Code requirement, but is

(Continued on page 53)

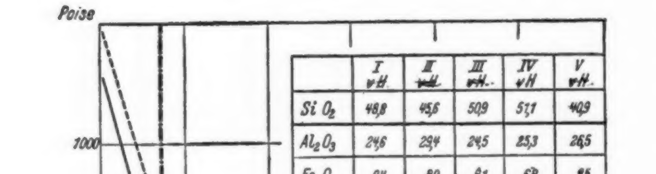
Influences Affecting Slag Behavior

This review of the slagging problem is based on German experience with high-ash coals, up to as much as 30 per cent. The influence of various components of the ash is discussed; and, although the conclusions differ in some respects from the results of investigations in this country, they afford an interesting comparison—Editor.

By DR. O. A. F. MÜNZINGER,
Consulting Engineer, Berlin

As MENTIONED in my article in August 1947 number of COMBUSTION, prior to and during World War II many German boiler plants were faced with the problem of burning large quantities of bituminous coal having an ash content up to 30 per cent; also coke from the low-temperature carbonization of brown coal. The ash resulting from the burning of these two fuels, even with moderate furnace temperatures, produced very hard deposits on the heating surfaces. Moreover, further difficulties arose when the first few large pulverized-coal-fire boilers of the slagging-bottom type were placed in service.

Due to the varying ash characteristics of German bituminous coals, it was seldom that one could be sure of the fusion temperature of the ash. Investigations of samples by means of the Bunte-Baum method were carried out, but in many cases the results were misleading due to the character of this and similar methods of investigation which only approximate actual furnace conditions.



	I vH	II vH	III vH	IV vH	V vH
SiO_2	48,8	45,6	50,9	51,1	40,9
Al_2O_3	24,6	29,4	24,5	25,3	26,5
FeO	9,4	8,9	8,1	6,8	9,5
Fe_2O_3	3,5	3,5	4,5	3,4	15,8
CaO	6,7	3,5	3,8	4,5	5,3
MgO	2,4	2,6	2,1	2,6	2,0
K_2O	2,4	4,6.	3,1	3,0	—
Na_2O	0,2	0,7	2,24	1,2	—
S	0,02	0,1		0,04	
$SiO_2 + Al_2O_3$	73,4	75,0	75,4	75,4	
$K_2 = 0.5 \cdot Fe + 0.5 \cdot CaO + MgO$	16,5	13,9	13,5	12,5	23,2

Fig. 1—Variation of viscosity with temperature for ash of different compositions

However, with the aid of the ultramicroscope, Dr. Zinson was able to accomplish much in establishing the relation between ash analysis and behavior of the slag. This instrument permits a rapid and close determination of the temperature at which an ash will show the first signs of altering its shape, the temperature at which it will begin to soften, and that at which it is completely molten. The results were plotted on three-coordinate diagrams with the three principal components, namely, clay ($SiO_2 + Al_2O_3 + MgO$), lime and sulphur ($CaO + SO_3$), and iron oxides ($Fe_2O_3 + FeO$), as ordinates and the temperatures indicated for a large number of coals. According to the compositions of the samples examined, the ash-softening temperatures, determined from such plotting, ranged between 1500 and 2000 F. It was found necessary to keep the temperatures of the furnace gases entering the superheater or boiler heating surfaces having closely spaced tubes at least 100 deg F below the softening values plotted for the given coal; otherwise bridging would occur.

For this reason channel-type superheaters[1] have proved advantageous when burning coals containing troublesome ash, and when employing highly superheated steam, because with such construction a considerable part of the superheater can be placed in gas temperatures of 1800 to 2000 F without danger of it being slagged or bridged, as would be the case with more closely spaced superheater tubes.

Similar progress was made by the late Professor Endell with reference to pulverized-coal-fired furnaces of the slagging-bottom type. In this he discriminated between what he termed "long" and "short" slags. The viscosity of the former decreases slowly with falling temperature, but they still remain fluid. On the other hand, the "short" slags are very fluid at high temperature but solidify almost instantly if the temperature falls below a certain value. The danger of rapid freezing of molten slag on the furnace bottom at light load is therefore great with them.

According to the findings of Professor Endell, the viscosity factor of an ash, K_2, is well represented by the expression

[1] The channel-type superheater consists essentially of groups of superheater elements mounted in widely spaced panels hung over the furnace. The combustion gases pass parallel to the tubes and only a small portion of the gas comes in contact with the tubes, the heat transfer being largely by radiation. Thus there is less opportunity for the deposit of slag.—Editor.

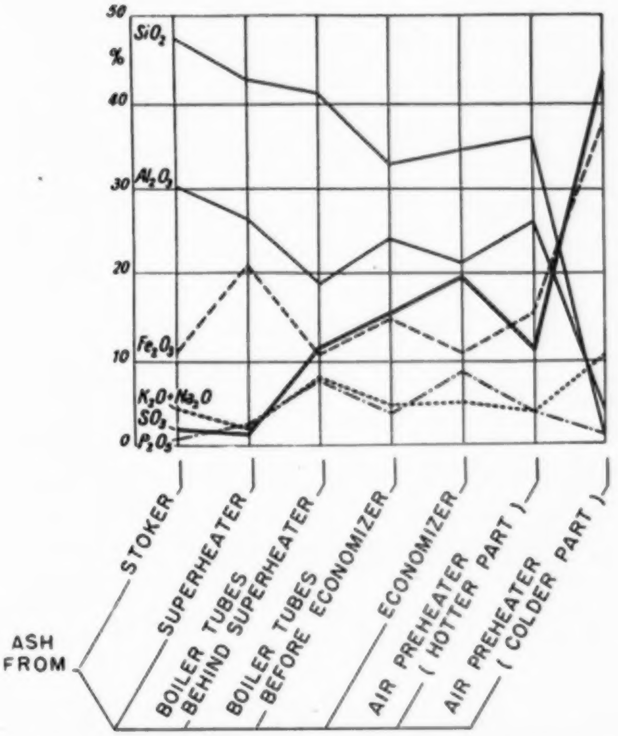


Fig. 2—Composition of ash deposits at various locations, ordinates represent percentages

$$K_2 = 0.5 \times Fe + 0.5 \times (CaO + MgO)$$

in which Fe, CaO and MgO represent the percentages by weight of the three most important constituents of the ash. It suffices to determine the content of Fe, CaO and MgO without having to resort to an investigation of the viscosity itself or to a determination of other constituents of the ash. For example, Curve V in Fig. 1 shows a slag, which, due to its high iron content, is typically short. At the lowest boiler load the furnace temperature must be at least 100 to 200 deg F higher than the temperature at which the molten slag has a viscosity of 100 to 500 poises. By way of comparison, to show what a viscosity of 100 to 500 poises means, it may be pointed out that summer automobile lubricating oil has a viscosity of around 100 poises at room temperature. By adding 5 per cent CaO or 2 per cent MgO the viscosity of the ash may be decreased from around 900 to 100 poises.

Investigations have shown that the formation of hard ash deposits on heating surfaces is brought about principally by two different causes. That on furnace water walls and superheaters is mainly due to reactions between certain ingredients—the so-called eutectica (calcium oxide, iron oxide, aluminum oxide and silicic acid) which according to the temperature and their mutual quantitative rates form slag coatings of different hardness.

On the other hand, hard coating on boiler and economizer surface is predominantly a process in which, under the influence of high temperature, complicated chemical reactions take place between sulphur, phosphoric acid and alkali. These tend particularly to foul the cooler parts of the heat-absorbing surfaces, such as later boiler passes, economizer and air heater.

Slagging, of course, may also be caused by delayed combustion. Incandescent coke particles entrained in the gases passing through a boiler will produce high local temperatures and the ash of the particles themselves or soft ash already deposited on the tubes will become molten and form a sticky mass. Furthermore, where the ash is high in iron and sulphur, incandescent coal particles floating in the gas stream have another bad effect, for at temperatures between 1400 and 1800 F they tend toward the formation of sulphides of calcium, iron and silicon, due to their reducing effect according to the following reactions:

$$SiO_2 + FeS + 2C = SiS + Fe + 2CO$$
$$FeS_2 + 2SiO_2 + 2C = 2SiS + Fe + 2CO_2$$

The sulphides sublimate (become gaseous) and when coming in contact with the relatively cool heating surfaces condense to form a sticky coating to which other ash particles adhere. As long as the flue gases contain excess air at temperatures above 700 to 900 F, the silicon sulphides adhering to the tubes will change as follows:

$$SiS + 2O_2 = SiO_2 + SO_2$$

The hard SiO_2 layers often found on boiler, superheater or economizer tubes are therefore not, as often assumed, produced by the early combustion stages but by the intermediate action of the sulfides and their burning to SiO_2 after deposition on the tubes.

Ash from some brown coals and from some low-temperature coke is especially troublesome because its gypsum content produces very hard layers of calcium sulphide and residual gypsum. If the ash contains considerable iron in the sulphide state, ferrous sulphate will result. This will disintegrate into iron oxide and SO_3, the latter corrosive upon passing over the cooler surfaces.

Fig. 2 shows the composition of ash deposits at different locations. At first the SiO_2 and the Al_2O_3 contents continuously decrease. The SO_3 at first remains nearly constant, then gradually increases and in the last part of the air heater suddenly increases rapidly, due to the combined effect of the distance covered by the flue gases and the decreasing temperature.

Only after complete combustion of the coal particles carried in the gases can the sulphides oxidize and lose their stickiness. Therefore, the heating surfaces will slag less the longer the path of the gases and the hotter the excess oxygen available for reaction with the sulphides.

Notwithstanding all precautionary measures, hard ash deposits cannot be avoided completely with some designs of boilers.

The formation of layers of ash and slag does not proceed linearly nor is it proportional to the boiler load. For some time the tubes remain clean until a thin layer containing silica has developed, and this slowly thickens. The fouling then increases with increasing temperature of the outside of the layer until the whole layer softens. Subsequently the fouling proceeds rapidly because the fly ash adheres to the plastic coating on the tubes.

The deduction to be drawn from the foregoing is that a furnace must be of such volume, such height and of such shape, that the particles of coal as well as the alkali oxides are totally burned before the gases leave it. The proper layout of a furnace is therefore indispensable both to high boiler efficiency and avoidance of excessive slagging of the heat-absorbing surfaces.

*Factors in Selection of Steam Generating Units**

There are many aspects of this problem that are not widely known or appreciated outside of circles actually engaged in such design, and little of a comprehensive nature has been published on the subject as pertaining to modern steam generators. The author differentiates the selection of small and large units and discusses the factors involved in each category. The procedure for large units is outlined step by step, and calculations are indicated. Some typical units are illustrated.

By W. S. Patterson

Combustion Engineering Co., Inc.

FACTORS affecting selection differ somewhat as between large and small steam generating units. Moreover, a size that would be considered large for one industry might be considered small for some other, such as the utility field. Therefore, for the purpose of the present discussion, the writer has taken 100,000 lb of steam per hour as the dividing line.

Small Steam Generating Units

Generally, purchase of equipment in this category will be based on the lowest cost that will satisfactorily meet the requirements. This, however, may involve a relatively expensive boiler which, because of its size or shape, minimizes the changes in an existing building.

Practically all small boilers are standardized with respect to details of construction, drum sizes, widths, tube lengths or drum centers; and the determination of heating surface simply involves use of a tabulation of the various dimensions.

Usually, the minimum cost boiler is obtained by employing the minimum furnace width that will satisfactorily supply the desired steam quantity, although the type of fuel-burning equipment may dictate the width. For example, the ideal arrangement for a traveling-grate stoker will be a long narrow grate, whereas for best results at the same capacity a spreader stoker should have a wider and shorter grate.

Furnace volume also is important in fitting a new unit into an existing space; hence the method of firing enters the picture in another way since stokers require less furnace volume than pulverized coal for the same fuel-burning rate.

Particularly in the case of small standardized boilers, the purchaser sometimes prefers equipment of the type previously installed, if its performance has been satisfactory. The reasons in such cases include familiarity of the operators with the equipment; the fact that it usually fits the space conditions; and that the same spare parts can be carried for new and old units.

Another factor bearing on boiler selection is draft conditions. If draft is to be furnished by an existing stack, there is obviously less latitude in selection than if an induced-draft fan is to be supplied.

The trend in recent years has been toward bent-tube boilers. This has been brought about partly by standardization, by improved fabricating methods that have reduced their cost; and because circulation in this type has proved superior to that in straight-tube boilers at high outputs, due to the steeper inclination of the tubes. Several manufacturers have developed standardized so-called "package" boilers of the low-head, bent-tube type, the three-drum low-head type and the two-drum vertical, bent-tube design.

Horizontal return-tubular boilers are usually lowest in cost, exclusive of setting, and are limited to about 20,000 lb of steam per hour, about 200 psi and 150 deg superheat. Next in order of cost is the three-drum, low-head, bent-tube design which can be built for capacities up to around 35,000 lb per hr, 400 psi pressure and 100 deg superheat. The box-header, straight-tube type, without water walls, has been built for capacities up to 75,000 lb per hr, pressures up to 300 psi and 150 deg superheat. However, this type has not been used extensively in recent years. Next comes the two-drum, vertical, bent-tube boiler with water-cooled furnace, built in capacities up to 300,000 lb per hr and for pressures up to 1000 psi with steam temperatures up to 900 F. Such units are adapted to all types of firing and many have been installed in both industrial and utility plants.

The three-drum, vertical, bent-tube type has been offered by some manufacturers in relatively small sizes but the author's company has developed it for public utility and large industrial plant use in various capacities up to a million pounds of steam per hour and for the full range of pressures and temperatures, as well as for all methods of firing.

The four-drum, vertical, bent-tube design has been built for capacities up to 300,000 lb per hr (with water walls); for pressures up to 1000 psi and 800 F steam temperature. Like the three-drum design, it is adaptable to all types of firing.

The sectional-header, straight-tube type, with water walls, has been built for various capacities, even up to

* This text is based on a talk originally given before a small group constituting the Power Generation Committee of the A.E.I.C., and was later expanded into a paper before the Technical Association of the Pulp and Paper Industry. It was briefed, in part, by the author at a Panel Discussion during the Annual A.S.M.E. Meeting at Atlantic City last December.

Fig. 1—Factors influencing basic requirements for large steam generating units

500,000 lb per hr with pressures up to 1200 psi and 900 F steam temperature. It is adapted to all types of firing, but in recent years has been superseded by the bent-tube type for high pressures.

Large Steam Generating Units

In selecting large units particular consideration must be given to certain basic requirements contained in the customer's specifications, or which must be determined before a satisfactory selection can be made. These are as follows:

1. Customer's preference.
2. Building limitations
3. Evaluations
4. Maximum capacity
5. Capacity range
6. Steam temperature
7. Steam temperature control range
8. Pressure
9. Feed temperature
10. Primary fuel
11. Auxiliary or future fuel
12. Nature of feedwater
13. Desired efficiency

The design factors which must be considered in relation to each of these items from the specifications are enumerated in Charts 1 to 13 of Fig. 1. These will be discussed briefly, in order.

1. As in the case of small units, customer's preferences are often based on the desire to obtain a unit (*a*) similar to one with which he has had satisfactory results; (*b*) of a type with which his operating personnel is familiar and (*c*) which fits well alongside existing units in the plant.

2. Building limitations must receive next consideration, particularly if the new unit is to be placed in a space provided by removing one or more old boilers. Often it is possible to install new units of several times the capacity of the old ones while at the same time using the existing coal bunker and providing an operating floor and fan floor at the same level as previously used. Preferred locations of pulverizers, fans, air heater and dust collector must, of course, be determined in advance.

3. Even before a preliminary selection is made the designer will want to consider *evaluations*. These are really monetary penalties that will be applied to the bid price in an attempt to put all bids on the same basis as regards space occupied, power consumption of auxiliaries, such as fans and pulverizers, and overall efficiency. If the desired efficiency can be obtained with low draft loss, it would not be wise to bid on a high draft loss unit unless the evaluation for saving in space would more than offset the increased cost of heating surface, etc., of the low draft loss unit.

4. Several design factors are related to capacity. One of these is type of firing. Wherever coal must be considered, a decision must be made between stokers and pulverized coal. However, with a few exceptions, stokers are limited to small and medium-size units, whereas pulverized coal is not economical when the size of unit is too small.

Furnace width is also related to capacity because it affects the fuel-burning rate, the draft loss and the effective steam drum length, as well as cost of the unit. The drum length and diameter fixes the volume which must be large enough to permit installation of the necessary steam purification apparatus. Boilers having a single upper drum are generally wider for the same capacity; but sometimes the fuel-burning equipment will fix the width rather than drum capacity.

5. Range of capacity will influence the type of boiler offered, particularly when used as a standby for a hydroelectric system. It will also affect the type of firing. For small or medium-size plants stokers are often favored over pulverized coal when frequent large load swings of the order of 10 to 1 must be met or when operation for long periods at greatly reduced load is necessary. However, stokers seldom are used for steam capacities greater than 200,000 lb per hr. For pulverized coal a greater range in capacity can be obtained with turbulent burners, although the characteristics of the pulverizers also play a part in direct-fired installations. Gas and oil firing are best suited to a wide range in capacity if burner selection is carefully made. Pulverized-coal-fired units designed for a wide range in capacity, and which must operate for long periods at low load, should have dry-ash-bottom furnaces rather than the slag-tap type.

6. The specified steam temperature influences the design of boiler inasmuch as it affects the size and location of the superheater. Some users of large units are now specifying 1000 F, 1050 F, or even 1100 F, and some are planning interstage reheating to 1000 F. It may be said of such units that the rest of the installation is really built around the superheater which may account for 40 per cent or more of the total heat absorption by pressure parts.

When the gas temperature entering the superheater must be close to the fusion temperature of the ash, it is customary to divide the superheater surface into two or more sections having a wide tube spacing in the gas inlet section and a much closer spacing in the gas outlet section.

7. Another basic requirement which influences steam generator design is the steam temperature control range; that is, the load range over which steam temperature must be held constant. All or any one of four means of control may be provided, such as adjustable burners, dampers, desuperheating, or furnace wall blowers.

When dampers are used the arrangement of the surface of boiler and superheater must be coordinated so as to bypass effectively a predetermined portion of the superheater surface; to cool the bypassed gas effectively; to provide a reasonably cool location for the dampers; and to facilitate the installation of a damper arrangement that will be positive and quick-acting in the control of superheat and adaptable to automatic controls.

The effect of steam temperature control range on gas temperature requirements at the furnace exit will be discussed in some detail later.

When spray-type desuperheaters are the sole means of temperature control they require a large quantity of water which has bypassed the economizer and is converted into superheated steam. The reduced flow through the economizer decreases economizer efficiency in the upper load range because the outlet water temperature will more rapidly approach the saturation temperature.

8. Pressure also governs several design factors, par-

ticularly the type of boiler. High pressure means a decreased specific volume of steam which makes possible fewer or smaller drums. But at high pressure the density of water and steam approach each other and thus reduce the head which produces thermal circulation. For this reason generous downtake and generating tube diameters must be employed. However, since tube diameter and wall thickness are roughly directly proportional, a point is reached where wall thickness becomes critical. This results in a large temperature gradient across the tube wall, a high temperature on the hot face of the tube and a high temperature stress in the tube wall. It is for this reason that high-pressure boilers are sometimes provided with refractory protection on the fire side of furnace wall tubes if the latter are very thick. Alternatives are smaller diameter tubes, or forced circulation.

Also, in high-pressure units the boiler-water concentration of dissolved solids is generally held to a lower figure than with low-pressure units by means of continuous blowdown. The blowdown rate must be considered at the time of boiler selection.

9. The next basic item is feedwater temperature which the designer uses to calculate the required total heat absorption of the unit and the maximum percentage of that total which could be absorbed in an economizer without steaming.

Feedwater temperature determines the metal temperature in an economizer which should be above the dew point of the flue gas, because the temperature of the gas film in contact with the metal will approach the metal temperature, regardless of the mean gas temperature. Since deaerated feedwater is generally employed for large boilers, the minimum feedwater temperature is usually 215 to 220 F, but a temperature as high as 580 F has actually been specified for some very high-pressure units by employing high-pressure extraction feed heaters.

10. Fuel analysis is a most important item of the specifications. This is particularly true when more than one fuel must be used, either separately or simultaneously. The various design factors related to coal analysis, as applied to pulverized coal, are listed in Chart 10, Fig. 1. For stoker firing additional items concerning the coal are desirable, such as sizing, iron oxide content of the ash, and burning characteristics.

11. The specified use of auxiliary or future fuel must also receive careful consideration, as it may affect the type of firing offered. Of all the fuels that may be specified, that requiring the largest furnace volume will fix the initial volume. Furnace bottom type must also receive special consideration. No unit will give the same performance with all fuels, and it is more difficult to design a unit for satisfactory operation with several fuels than with one. Unless superheat control is furnished or superheater alterations are made when changing from one fuel to another, a different steam temperature may be obtained with each fuel.

12. The nature of the feedwater governs the factors listed in Chart 12, some of which are of particular interest to the water consultant and others to the boiler designer.

13. When the desired efficiency is specified, it is not too difficult to arrive at a trial value of exit gas temperature which is then used as a guide in equipment selection. Most large boilers designed for an overall efficiency of 85 per cent, or higher, will require heat-recovery equipment including both economizer and air heater. If the unit is stoker-fired, the air heater will be small and the economizer large, but the reverse is true for a pulverized-fuel installation. With units designed for only moderate efficiency, the air heater is usually omitted for stoker firing and the economizer omitted for pulverized-coal firing. Gas- and oil-fired units will have heat-recovery equipment proportioned about the same as for pulverized coal, because from the standpoint of economics a large air heater and small economizer are cheaper than the reverse combination.

When the desired efficiency results in a gas temperature close to the dew point at normal full load, as will often be the case with large high-pressure units, means are generally provided to control the cold-end metal temperature of the air heater by recirculating preheated air or passing cold air.

Calculations Involved in Equipment Selection

(*a*) Capacity, pressure and steam temperature determine the heat in the steam leaving the unit, from which the heat in the feed is subtracted to arrive at the heat output of the unit at various loads such as minimum, "control point" and maximum.

(*b*) Experienced judgment plus information from the specifications are then used to assume the efficiency at each load and calculate the required heat input in the fuel.

(*c*) From the heat input and excess air the gas weight may then be calculated.

(*d*) It may be said of large, high-temperature steam generating units that the superheater requirements to a large extent determine the type of unit. This is true to a certain extent in the 900 F to 1000 F designs, illustrated in Figs. 3 and 4, but to a much larger extent when higher steam temperatures are required or when there is both high superheat and high reheat as in Fig. 5.

Fig. 2 illustrates the relationship between steam temperature rise and gas temperature drop in a convection superheater. The steam temperature rise ($T_2 - T_1$) and the pressure fix the heat to be absorbed per pound of steam and this may be specified constant over a wide range of loads. If the feedwater temperature and excess air are nearly constant then the gas weight per pound of

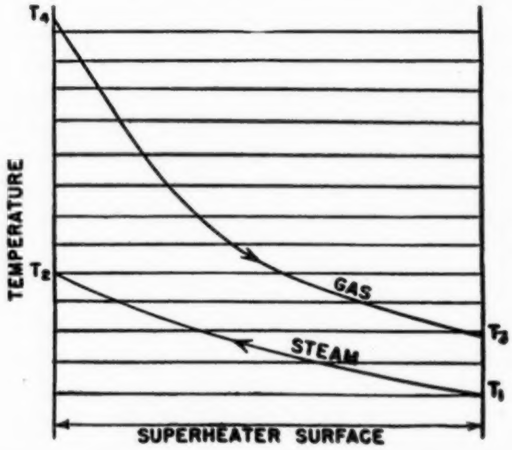


Fig. 2—Relation between steam temperature and gas temperature drop for convection superheater

steam and the gas temperature drop ($T_4 - T_3$) will also be nearly constant over the superheat control range.

The amount of surface installed in the superheater, and the degree to which true counterflow is approached, will determine the cold end temperature difference ($T_3 - T_1$). A counterflow arrangement will economically give a gas temperature drop ($T_4 - T_3$) equal to 85 per cent of the temperature difference ($T_4 - T_1$) at the design or "control" point when the control point is at 50 per cent of maximum load. The absorption efficiency will then fall off to 75 per cent at maximum load due to the fact that the heat transfer rate in the superheater does not increase in direct proportion to the increase in gas weight. Therefore, the entering gas temperature T_4 required to maintain constant steam temperature T_2 will increase only moderately with rating when the feedwater temperature remains constant.

For example, the gas temperature drop ($T_4 - T_3$) in a convection superheater, corresponding to a 400 F steam temperature rise ($T_2 - T_1$), will be about 1025 F and the temperature difference ($T_4 - T_1$) may be taken as 1025/0.85 = 1200 F (approximately) at the control point and 1025/0.75 = 1365 F at the maximum load for 50 per cent control range. These figures fix the required gas temperature entering the superheater at 1800 F for the control point and 1965 F at the maximum load when the saturated steam temperature is taken as 600 F. (The

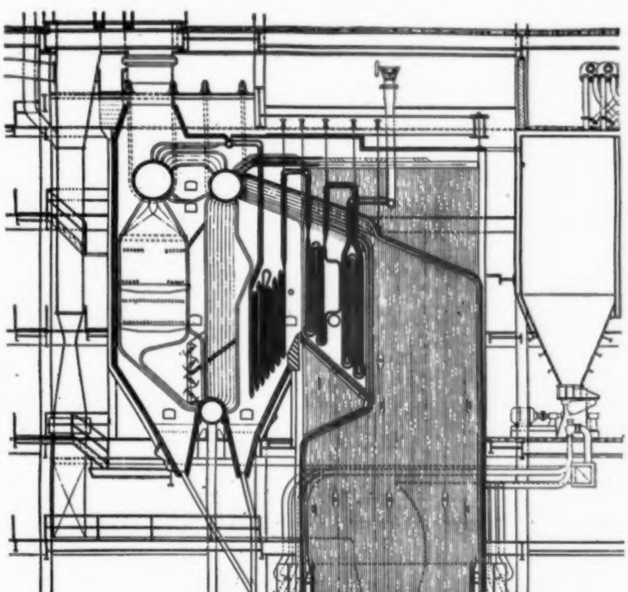


Fig. 3—A recent 600,000-lb per hr, 1380-psi, 950-F pulverized-coal-fired unit

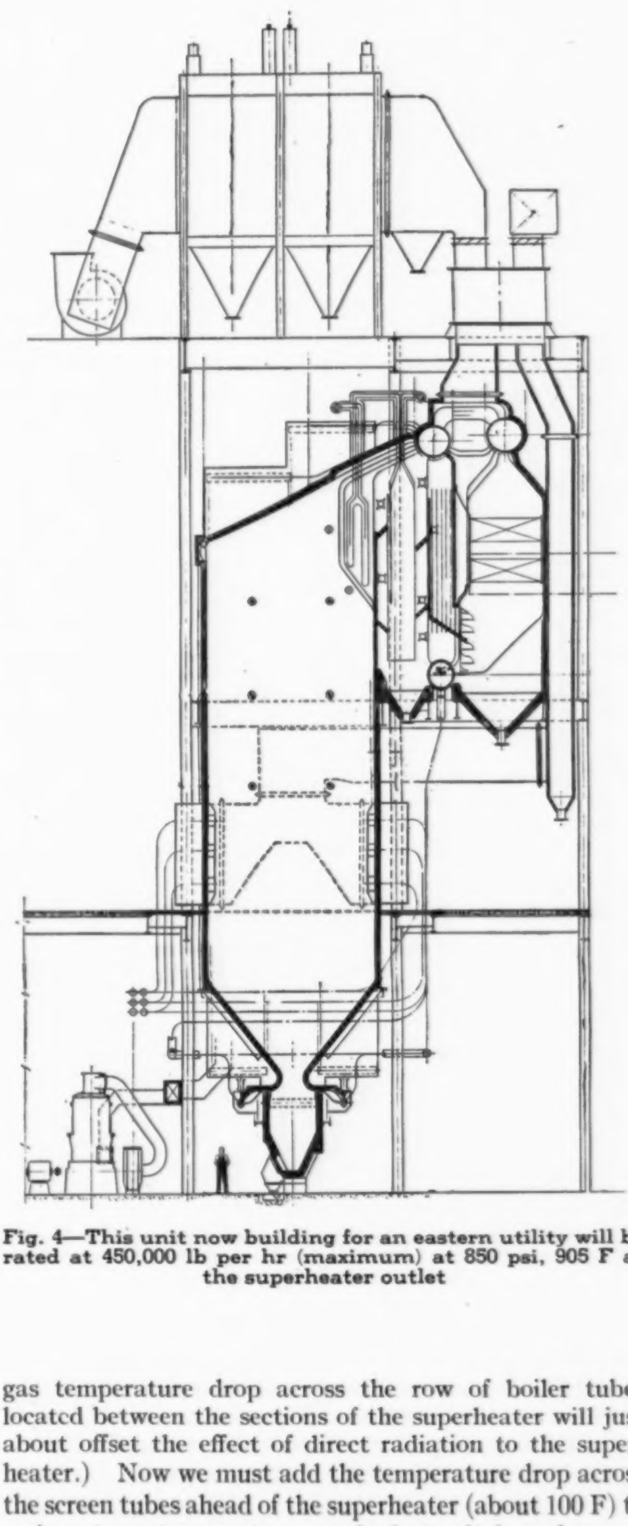


Fig. 4—This unit now building for an eastern utility will be rated at 450,000 lb per hr (maximum) at 850 psi, 905 F at the superheater outlet

gas temperature drop across the row of boiler tubes located between the sections of the superheater will just about offset the effect of direct radiation to the superheater.) Now we must add the temperature drop across the screen tubes ahead of the superheater (about 100 F) to arrive at gas temperature required at exit from furnace, i.e., 1900 F at control point and 2065 F at maximum load.

A furnace must therefore be selected to give an exit temperature of 1900 F at 50 per cent of maximum load. A selection based on fixed tangential burners or horizontal turbulent burners would give trouble at maximum load because the furnace exit temperature would be about 2150 F to 2300 F, respectively, which would exceed the ash-fusion temperature of some coals and greatly exceed the temperature required to obtain the specified steam temperature. An alternative with fixed burners is to place part of the superheater in the furnace walls or in the walls of an open pass where temperatures above the fusion

point can be tolerated. This will reduce the amount of convection superheater required and will therefore reduce the required gas temperature entering that surface.

With tilting burners, which in effect give an adjustable furnace, the solution is simpler and more flexible. The furnace can be designed to give the required 1900 F at the control point with burners horizontal and they are tilted downward at higher loads to limit the gas temperature increase to exactly that required to give the constant steam temperature.

The units shown in Figs. 3 and 4 also employ bypass dampers as a supplementary means of steam temperature control. Specifications often impose an emergency condition of low feedwater temperature when some of the feedwater heaters are out of service. To maintain constant steam temperature at high loads with lower feedwater temperature is equivalent to a large increase in the control range and supplementary steam temperature control is necessary or desirable under these conditions. The extent to which the bypass dampers will be used is checked by calculating the gas temperature requirements at the superheater inlet under normal and emergency maximum load conditions. If the gas temperature control requirements exceed the control range obtainable with tilting burners then the bypass dampers are relied upon and the design calculations take this into consideration. The maximum range of control can be obtained by selecting the furnace large enough to require partial upward tilt of the burners at the control point, and maximum downward tilt at maximum load supplemented by some other means of control.

(*e*) Having decided on a convection superheater, tilting tangential burners and required gas temperature leaving the furnace at the control point, the next step is to proportion the furnace to accommodate the burners and to give the desired leaving temperature. The width is to a certain extent fixed by the capacity of the unit. The amount of variation permitted depends on type of unit but the width chosen may affect the arrangement of superheater and economizer because of gas velocity considerations.

The furnace width is related also to drum-relieving capacity except with special designs where the drum length can be chosen independent of furnace width within reasonable limits. The depth is fixed largely by the burners because certain width-to-depth relationships are also maintained with tangential firing. The height must be chosen with consideration to the burners. With tangential firing suitable distance is maintained between the upper burners and the entrance to the superheater and between the lower burners and the hopper. The final proportions must, of course, suit the building limitations and provide a furnace having the necessary water-cooled surface to give the desired gas temperature entering the superheater at the control point.

The designs shown both employ a water-cooled hopper-bottom furnace because the range of furnace temperature control is greatest when a considerable portion of the furnace heating surface is located below the level of the burners.

An empirical method of determining furnace heat absorption is used. This method is based on numerous tests over a period of years on steam generating units of different sizes, having different methods of firing and operating at variable rating under different conditions of furnace cleanliness. The results have been correlated by plotting furnace heat absorption rate against release rate with the method of firing, furnace cleanliness and excess air as parameters. With the advent of tilting burners, the angle of tilt has also been taken into consideration and numerous tests have been conducted to establish the effect of burner adjustments.

(*f*) Superheater selection and arrangement must satisfy many other requirements besides absorbing the necessary heat. The tube diameter and circuit length must be proportioned to suit the allowable pressure drop; the element spacing in a direction transverse to the gas flow must be satisfactory for the gas temperature, velocity and ash content; the free area must be chosen to suit the draft loss; the depth spacing must not impose fabrication difficulties; the circuit arangement must be chosen so as to insure an efficient temperature difference between gas and superheater surface in all parts of the superheater because this affects the amount of surface installed; and, finally, the materials chosen must be suitable for the metal temperature expected which, in turn, is a function not only of the fluid temperature inside the tube but also of the rate of heat absorption by the surface.

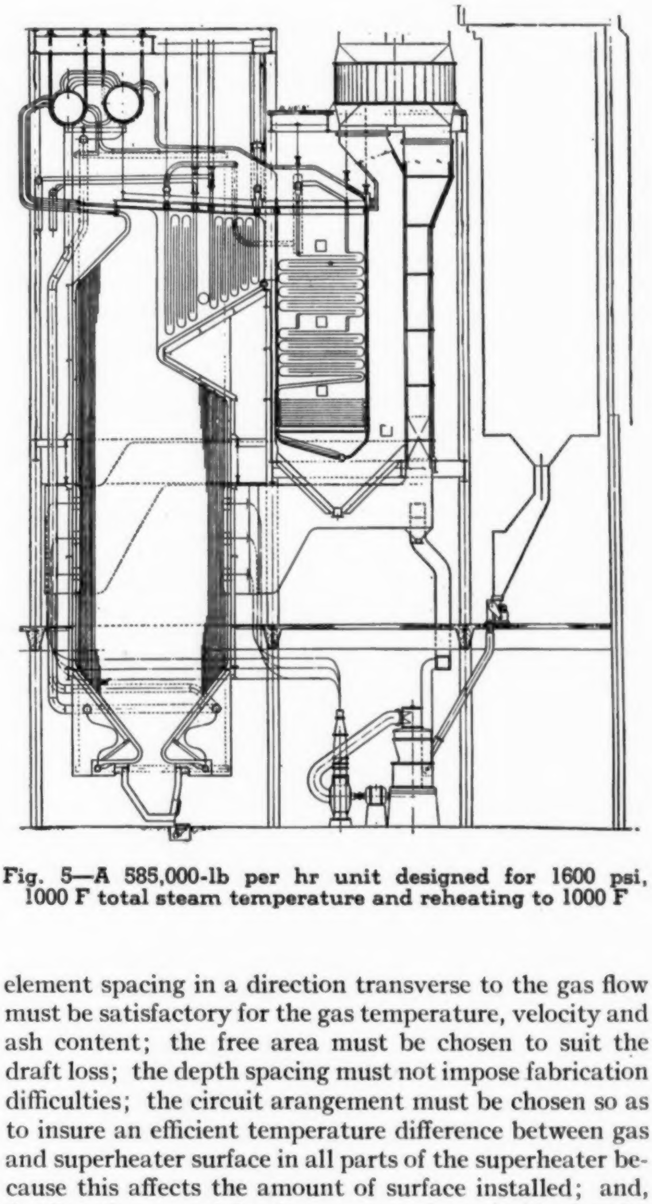


Fig. 5—A 585,000-lb per hr unit designed for 1600 psi, 1000 F total steam temperature and reheating to 1000 F

Modern high-capacity units invariably employ two or more transverse spacing arrangements and two or more superheater sections. The section nearest the furnace has wider spaced elements, lower gas velocity and higher steam velocity than the cold-end section. Because the temperature difference is great the section nearest the furnace is often arranged for parallel instead of counter

flow to reduce the maximum metal temperature. This does not add to the cost of the superheater for, although the required surface may be slightly greater, less of the more expensive alloy is required.

The units shown in Figs. 3 and 4 employ pendant type elements with all load-carrying supports located outside the setting. The latter has a counterflow circuit arrangement and transverse flow of gas over the tubes. The absence of baffles and zones of high gas velocity are outstanding features.

(*g*) In most modern, high-capacity designs there is a minimum of boiler surface ahead of the superheater. For structural reasons some of the rear water-wall tubes are located at the front and some at the rear of the front superheater section. Boiler surface beyond the superheater is also sometimes a matter of structural convenience in providing water-cooled supports for baffles, dampers and economizers. When bypass dampers are used for superheat control, it is desirable to arrange the baffles so that most of the boiler surface is in the bypass. This improves the efficiency at high loads, lowers the temperature at the dampers and results in more uniform temperature distribution entering the economizer. Where very high superheat, or both high superheat and reheat are involved, the design becomes special and such units are characterized by absence of boiler surface beyond the superheater (Fig. 5).

(*h*) Now we skip over the economizer to the air preheater because this piece of equipment is generally selected before the economizer. This procedure is necessary because it is the function of the air preheater not only to cool the gas to the temperature required to meet the desired overall efficiency but also to preheat the air to the temperature required for efficient drying in the pulverizer. Therefore, the coal specifications have a lot to do with selection of air heater size in tending to fix minimum gas temperature (with consideration to condensation of acid in the heater) and required air temperature. The heat absorption required by the air fixes the gas temperature drop in the heater and thus fixes the required gas temperature leaving the economizer. The heat absorption of the heater more or less fixes the length of the tubes, plates or sheets but the allowable draft loss and pressure drop fix the actual physical size or plan area.

(*i*) The economizer is selected to absorb the heat necessary to cool the gas from boiler outlet temperature to air heater inlet temperature. Generally, the superheater and air heater absorb such a large percentage of the total that the water does not closely approach the saturation temperature in the economizer. It is considered desirable not to approach saturation temperature closer than 35 deg F in the economizer.

(*j*) The number of pulverizers required for large units is generally three or four. Often the specifications call for full output to be carried by the remaining pulverizers when one is out of service for repairs. This requirement results in oversize pulverizers or a larger number. The size and total capacity must also be based on minimum grindability, minimum heat value and maximum moisture content, and if these requirements are partly conjectural or extreme emergency conditions it is sometimes questionable whether it is desirable to select pulverizers large enough to allow for the simultaneous occurrence of these conditions because oversize pulverizers will limit the minimum load on the unit.

EQUIPMENT SALES

as reported by equipment manufacturers to the Department of Commerce, Bureau of the Census

Boiler Sales

Stationary Power Boilers

	1947 Water Tube		1946 Water Tube		1947 Fire Tube		1946 Fire Tube	
	No.	Sq Ft*	No.	Sq Ft*	No.	Sq Ft*	No.	Sq Ft*
Jan.	160	963,949	173	1,109,924	106	106,788	113	154,064
Feb.	149	969,541	197	1,262,520	90	99,267	126	171,100
Mar.	168	863,292	171	1,357,650	80	109,984	123	180,552
Apr.	177	1,032,452	198	1,247,693	71	94,315	110	137,614
May	150	988,794	158	980,004	56	73,821	86	117,554
June	176	1,543,479	151	980,231	49	68,491	99	157,664
July	147†	1,017,090†	191	1,469,638	85	99,628	88	97,276
Aug.	153†	1,002,002†	128	877,497	60	78,470	117	147,684
Sept.	113	711,949	136†	1,205,364†	71	86,999	73	94,164
Oct.	152†	724,329†	187	1,591,936	87	80,191	120	148,768
Nov.	152	916,968	150†	1,036,978†	32	45,559	73	70,332
Dec.	171	959,372	157	1,040,976	83	68,534	81	79,804
Jan.–Dec., incl.	1,697	11,693,217	1,997	14,160,411	870	1,012,047	1,219	1,556,576

* Includes water wall heating surface. † Revised.

Total steam generating capacity of water tube boilers during the period Jan. to Dec. (incl.) 1947, 128,646,000 lb per hr; in 1946, 136,547,000 lb per hr.

Marine Boiler Sales

	1947 Water Tube		1946 Water Tube		1947 Scotch		1946 Scotch	
	No.	Sq Ft*	No.	Sq. Ft*	No.	Sq Ft	No.	Sq Ft*
Jan.	2	7,724	2	11,276	—	——	1	590
Feb.	2	1,423	—	——	—	——	—	
Mar.	5	22,232	—	——	—	——	4	1,706
Apr.	11	6,801	18	46,390	—	——	1	263
May	1	4,852	4	9,040	—	——	1	263
June	1	688	31	17,620	1	1,290	1	520
July	4	25,834	2	7,424	—	——	3	539
Aug.	3	14,160	—	——	—	——	16	5,087
Sept.	2	12,446	5	11,836	—	——	1	990
Oct.	—	——	—	——	—	723	—	——
Nov.	—	——	—	——	—	——	—	——
Dec.	2	11,440	4	7,550	—	——	2	611
Jan.–Dec., incl.	33	107,600	66	111,136	2	2,013	30	10,569

* Includes water wall heating surface.

Total steam generating capacity of water tube boilers sold in the period Jan. to Dec. (incl.) 1947, 1,197,000 lb per hr; in 1946, 1,023,000 lb per hr.

††Mechanical Stoker Sales

	1947 Water Tube		1946 Water Tube		1947 Fire Tube		1946 Fire Tube	
	No.	Hp	No.	Hp	No.	Hp	No.	Hp
Jan.	67	32,532	61	35,757	148	22,320	184	23,323
Feb.	55	32,759	71	40,880	122	19,946	175	27,708
Mar.	55	26,956	94	45,646	225	29,705	181	28,071
Apr.	63	37,914	93	45,606	111	19,649	249	42,271
May	77	40,481	101	49,653	93	12,500	202	30,933
June	83	38,204	76	42,259	187	24,964	233	32,815
July	74	38,580	97	50,668	306	42,689	233	33,290
Aug.	97	55,356	72	30,101	301	42,396	355	40,716
Sept.	73	39,219	49	28,118	372	41,421	401	34,937
Oct.	48	20,561	77	36,320	225	31,962	377	42,134
Nov.	49	28,400	64	34,787	159	23,203	293	23,708
Dec.	68	28,091	51	28,435	162	22,855	288	21,468
Jan.–Dec., incl.	809	419,053	906	468,230	2,411	333,610	3,171	381,374

†† Capacity over 300 lb of coal per hour.

Energy Sources and Output by Electric Utilities

A REPORT just issued by the Federal Power Commission contains interesting charts showing at a glance the energy sources of electric utility production by geographic divisions, the comparative outputs by water power, coal, oil and natural gas, from 1920 on, as well as the percentage of total generation by source.

The data are based on monthly reports from all electric utilities in the United States, both privately and publicly owned. In all cases the kilowatt-hour production represents a summation of the net station outputs after deduction of energy used in the operation of auxiliary equipment within the generating plants.

Fig. 1 shows the comparative outputs of electric energy for public use in the various geographical divisions of the country, particularly with reference to the source. This chart is based on figures for 1946.

It will be noted that the Pacific, Mountain and East South Central sections developed the major part of their electric energy by water power. Important, though lesser, amounts were derived from water in New England, the Middle Atlantic states and the South Atlantic division. Each of these latter three regions, however, produced at least two-thirds of the electricity from fuels and in each of these regions the major fuel was coal. The East North Central, West North Central, and West South Central regions are shown as predominately fuel-burning areas. Coal in the East North Central and gas in the West South Central group were the principal fuels in these respective areas. Coal accounted for about half the fuel-produced energy in the West North Central section, where gas and oil together supplied the other half.

This report shows that despite a 6½-fold increase in total output since 1920, the proportionate average outputs by coal, oil, gas and water power have changed very little. The average station fuel rate of 1.29 lbs per kw-hr for the whole country has remained nearly constant for several years, but varies geographically due to the character of coal available to different regions.

The steady increase in output by electric utilities since 1920, and the sources, are indicated in Fig. 2; whereas Fig. 3 shows that over the intervening years the proportionate outputs from the several sources have changed very little. For instance, while the amount produced by hydro has varied between 30.5 and 40 per cent, the average for the past five years has been about 35 per cent, despite

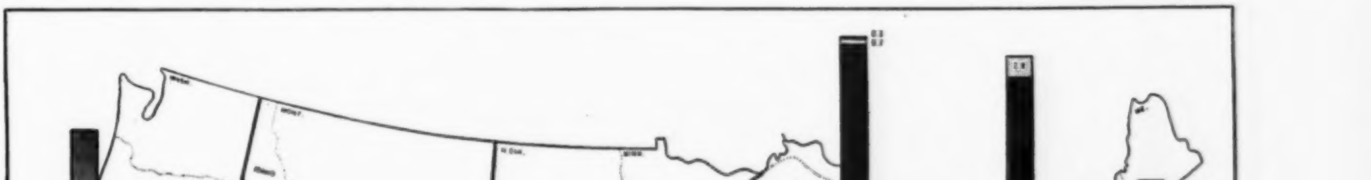


Fig. 1— Energy source of electric utility output by geographic divisions, 1946

the completion of large federally owned and operated hydro projects. For the year 1947 only 30.7 per cent of the total was hydro.

The electric energy production by utilities from all sources increased from 39.4 billion kilowatt-hours in 1920 to 255.7 billion kilowatt-hours in 1947—an increase of 12.1 per cent over that of 1946. This represented a 6.5-fold expansion during a period of 27 yr, or an average of 24 per cent per year. Fuel-burning plants accounted for 69.3 per cent of the 1947 output and hydro plants 30.7 per cent.

Large Fuel Gains in 1947

The report contains data on the amounts of coal, oil and natural gas consumed under utility boilers over these years. In 1947 the consumption of fuel by electric utility plants amounted to approximately $89^1/_2$ million tons of coal (a gain of 24 per cent over 1946); 45.3 million barrels of oil (an increase of 24.7 per cent over 1946); and 373,168,761 mcf of natural gas (a 21.6 per cent increase over the preceding year).

The annual average fuel rate per kilowatt-hour, for the county as a whole, based on coal and coal equivalents, decreased from 3 lb in 1920 to 1.29 in 1947. This figure has changed very little over the past five years, but varies with geographical location, according to the quality of coals burned. The range was from 1.11 lb per kw-hr in the South Atlantic region to 2.7 lb in the Mountain region. Breaking down the locations further, the best showing was in the District of Columbia with 0.98 lb per kw-hr, followed closely by New Jersey with 1.05 lb and Michigan with 1.07 lb. Of course, certain individual plants have consistently made still better showings.

In addition to the foregoing information on utility power, the report also contains data on electricity generated by private industrial plants. This was based on reports from approximately 800 plants which account for 85 per cent of the total industrial production of electricity, the figures being extended to represent 100 per cent coverage. These showed that such plants produced over 51 billion kilowatt-hours in 1947, with an installed capacity of about 13 million kilowatts. This compares with an installed capacity of 52,211,488 kilowatts in electric utility plants (private and publicly owned) at the end of 1947. Of the latter figure approximately ten million kilowatts represented federal and municipal plants.

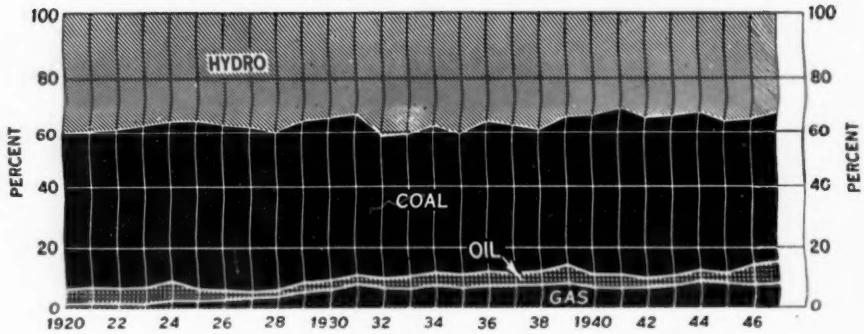


Fig. 3—Per cent of total generation by source of energy

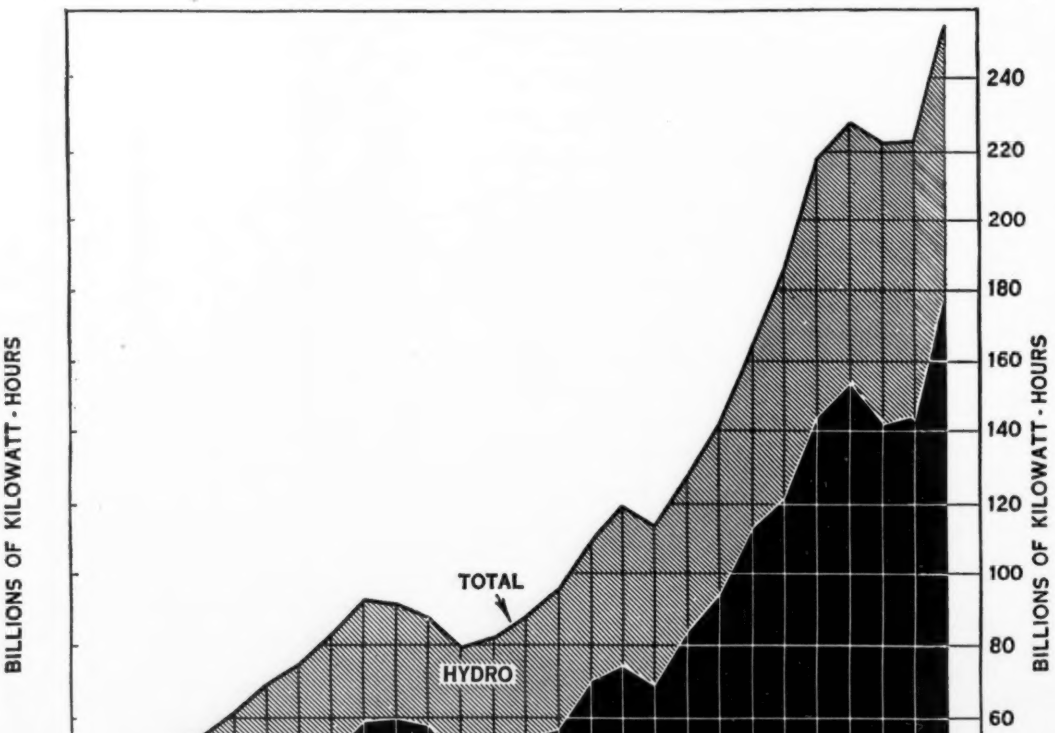
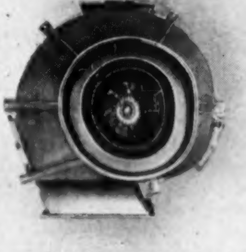
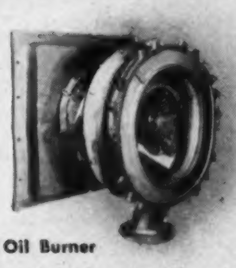


Fig 2—Growth of electric utility output by sources of energy, 1920–1947

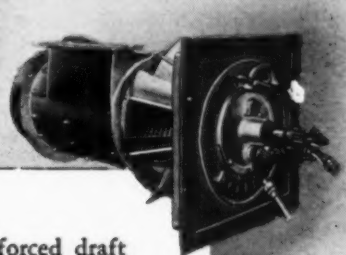


Blast Furnace Gas Burner

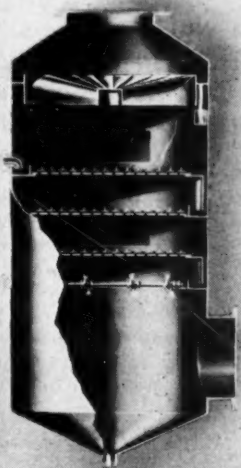
Petroleum Refinery Burner for Gas and Oil



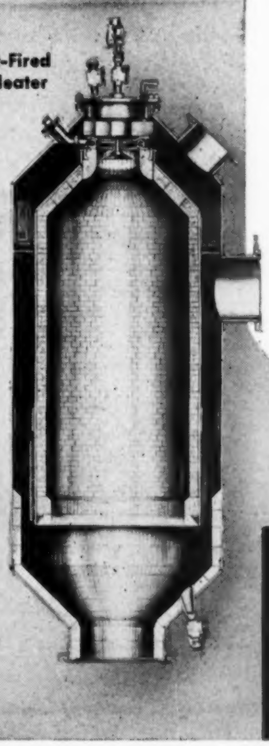
Combined Gas and Oil Burner



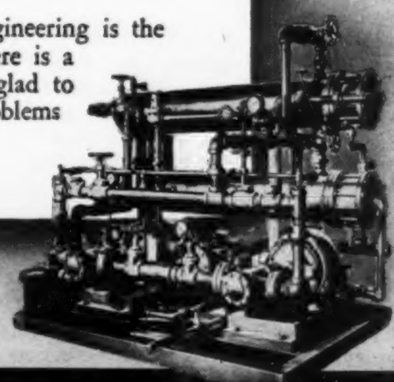
Combined Coal and Oil Burner



Gas Scrubbers, Coolers, Absorbers



Direct-Fired Air Heater



Fuel Oil Pump and Heater Set

GAS TURBINE HISTORY

In delivering the twentieth Thomas Lowe Gray lecture before The Institution of Mechanical Engineers, in London on January 23, Mr. T. A. Crowe chose as a subject, "The Gas Turbine as Applied to Marine Propulsion." The introduction to his paper consisted of a review of the historical development of the gas turbine and brief comments on the open, closed and semi-closed cycles. In view of the prominence attached to this type of prime mover within the last few years, it is believed that readers will be interested in what Mr. Crowe has to say of its historical development; hence the following excerpts are offered, without attempting to include that part of the paper dealing particularly with marine applications.

DEVELOPMENT of the turbine gas has been proceeding over a long period. The first patent was granted in 1791 to John Barber, whose proposals were surprisingly close to modern principles.

Coal or wood gas from retorts, and a supply of air, were pumped into receivers where the pressure was kept steady by the head of water in an overhead tank. The gas and air then flowed into the combustion chamber where a jet of water was added, and the resulting combustion gases impinged on the vanes of the turbine wheel. This was to provide the power to drive the pumps through S-shaped cams, and also, it was hoped, to drive power machinery.

Early in the nineteenth century Stirling and Ericsson both experimented with hot-air engines, but these were of the reciprocating type, and not turbines.

Stirling's engine, patented in 1827, was the first to incorporate a regenerator, or heat-exchanger, which plays such an important part in modern gas-turbine cycles. In the regenerator, the hot exhaust gases from the engine were used to preheat the compressed air before it entered the air heater proper. This regeneration took place while the air was maintained at constant volume, so that the pressure increased with the rise of temperature. Regeneration at constant volume can be adopted only in reciprocating engines. It was Ericsson who later suggested that it should take place at constant pressure, a procedure which is suitable for both turbines and reciprocating engines.

The pressure-volume and temperature-entropy diagrams for both the Stirling (*a*) and the Ericsson (*b*) cycles are shown in Fig. 1. The entropy diagrams for the two cycles differ only in their slopes, a line of constant volume on the diagram being steeper than a line of constant pressure. Both these cycles have a thermal efficiency equal to the Carnot efficiency, but, as in the Carnot cycle, both involve isothermal compression and expansion which is impossible to realize in practical machines. In actual plants, compression and expansion take place approximately adiabatically. In 1851 Joule proposed such a cycle as in Fig. 2 (*a*).

The Joule cycle is the basic cycle of all constant-pressure gas turbines, and has an efficiency lower than the Carnot cycle operating between the same temperatures. Fig. 2 (*b*) shows how by compressing adiabatically in a number of stages and cooling to the original temperature between each stage, and likewise expanding in stages with intermediate reheating, a closer approach may be made to the Ericsson cycle, and hence to the Carnot efficiency.

In 1861 a patent was granted to M. A. F. Mennons for an open-cycle gas turbine which was to operate with solid fuel. This incorporated all the principal components of a modern open-cycle unit, including a regenerator. Fig. 3 is taken from his patent specification. At the right-hand end of the shaft, a single-stage centrifugal impeller compresses the air and passes it through the inner passages of the regenerator, which is seen more clearly in the section at the right. The preheated air passes to the furnaces where solid fuel is burned and the "dilated" combustion gases pass out radially through the turbine impeller, and then flow back toward the right, through the outer heat-exchanger passages, and are discharged to the atmosphere.

The Curtis Patents

Charles G. Curtis, who did notable pioneer work in the field of steam turbines in the United States, took out a patent in 1895 for an open-cycle gas turbine using solid, liquid or gaseous fuel. He advocated the use of a multi-stage, axial-flow compressor, a velocity-compounded impulse turbine, and suggested water cooling of the nozzles, rotor and stator. Fig. 4 represents one arrangement proposed by Curtis, using fuel oil.

Despite all this early work, little success was obtained in the practical development of the gas turbine, chiefly because of the materials situation and the low efficiencies of both compressors and turbines. That is, the compressor absorbed more power than was developed by the turbine. However, in the early days, little was known about the creep of steel at high temperatures.

The first large-scale gas turbine to develop sufficient power to drive its own compressor and still have a margin to do useful work, was built by the Société des Turbomoteurs of Paris about 1905. This operated on the constant-pressure cycle and consisted of a Curtis wheel driving a 400-hp Rateau compressor of 25 stages in three cylinders. It ran at 4250 rpm and most of the compressed air was used for combus-

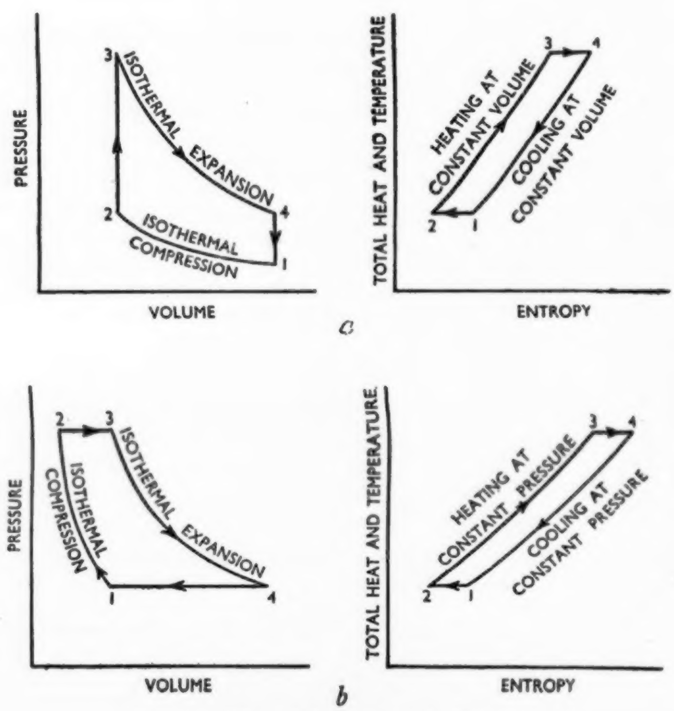


Fig. 1—The Stirling and Ericsson cycles

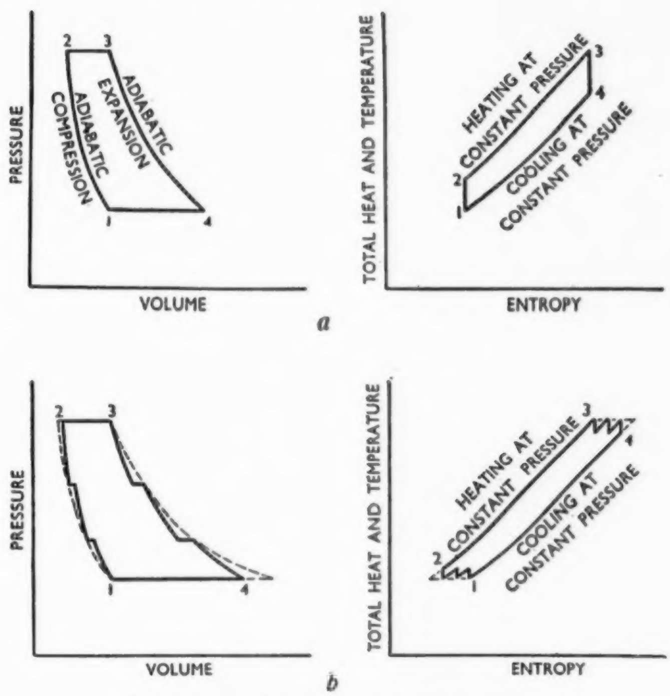


Fig. 2—The Joule cycle (*a*) and a cycle (*b*) approaching the Ericsson cycle by intercooling and reheating

tion and for cooling the turbine. The remaining air was tapped off for use elsewhere and represented the useful output of the unit. The thermal efficiency was about 3 per cent.

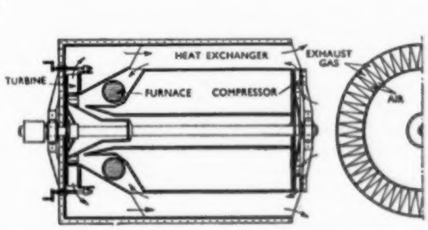


Fig. 3—Mennon's patent of 1861

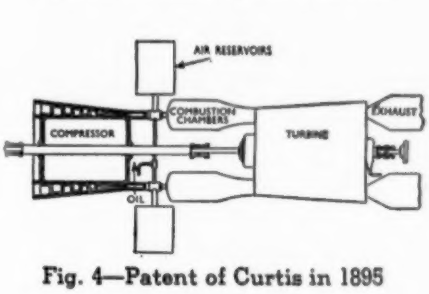


Fig. 4—Patent of Curtis in 1895

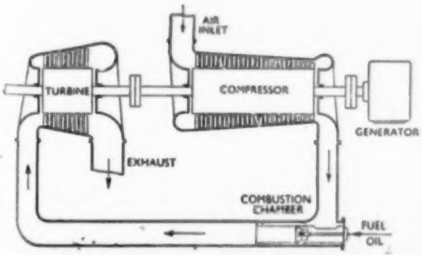


Fig. 5—Simple constant-pressure gas turbine

Holzwarth Designs

The inefficiency of early compressors caused designers to investigate cycles that avoided use of air compressors. However, the only cycle to offer a practical solution was the constant-volume cycle developed by Holzwarth around 1906.

In his original design air was taken into the combustion chamber at atmospheric pressure, after which the gas was admitted and fired electrically at constant volume, thus raising the pressure to between 70 and 100 psi. After ignition had taken place a valve opened and allowed the gas to pass through the nozzles to the turbine wheel which consisted of two velocity stages. The combustion chamber, nozzles and turbine were all water cooled, and the steam generated in the water jackets was used in a separate turbine which drove the scavenging blower.

The first Holzwarth gas turbine was built in 1908 by Körting at Hanover and was designed for 50 hp. A second Holzwarth machine, of vertical type, was built two years later by Brown Boveri and was designed for an output of 1000 hp. Although this output was never attained, the unit was considered to be the first practical gas turbine.

Much further development was done by Holzwarth and in later units the explosive mixture was precompressed to about 40 psi whereby the final explosion pressure was raised to about 200 psi. Two discharge valves were fitted to the explosion chamber, the second opening after the first, when the pressure had fallen considerably, and admitting the lower pressure gases to the turbine through a passage which bypassed the first turbine wheel. A 2000-hp Holzwarth unit, burning blast-furnace gas, was built by Brown Boveri and installed in the Thyssen Steel Works at Hamborn in 1933. During the war a 5000-hp unit was installed in Germany, but details are not available.

Although the constant-volume type of gas turbine developed by Holzwarth is attractive in that it dispenses with the need for an air compressor, the complication of the valve gear and the cooling system seem to outweigh this advantage. Therefore, the simpler form of constant-pressure gas turbine is generally preferred at present.

Constant-Pressure Turbine

The simplest form of constant-pressure gas turbine is shown diagrammatically in Fig. 5. Its essential features are: (1) an air compressor which compresses free air to three or four atmospheres; (2) a combustion chamber in which the fuel is burned and the gas raised to high temperature at constant pressure; and (3) a turbine in which the gas is expanded down to approximately atmospheric pressure.

The thermal efficiency of a turbine working on this simple cycle is low and varies with the temperatures and efficiencies of turbine and compressor, although the overall efficiency is more susceptible to that of the turbine than to that of the compressor. With present-day materials the thermal efficiency is of the order of 16 to 22 per cent.

Efficiency of a simple open-cycle turbine without a heat-exchanger also varies greatly with the pressure ratio adopted.

However, the efficiency of a gas turbine working on the simple constant-pressure cycle can be increased in various ways, the

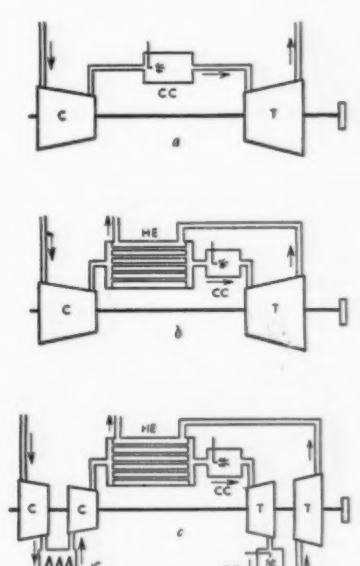


Fig. 6—Constant-pressure open-cycle arrangements for different efficiencies

greatest individual improvement being the addition of a heat-exchanger, by means of which the heat in the exhaust gases can be transmitted to the air between the compressor and the combustion chamber. The efficiency can be further improved by compounding the compressor, and applying intercooling between stages; also by compounding the turbine and reheating between stages. The effect of such inter-

cooling and reheating is to prevent excessive temperature changes, so that efficiency does not fall off at large pressure ratios.

Fig. 6 represents diagrammatically three open gas-turbine cycles with corresponding average efficiencies. A simple open-cycle gas turbine with 19.5 per cent efficiency is represented in the upper sketch, *a*. The middle sketch, *b*, shows such a turbine with a heat-exchanger added to bring the efficiency up to 27.2 per

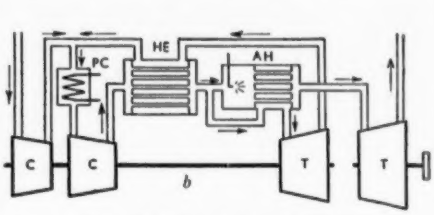


Fig. 7—Semi-closed cycle

cent. The bottom sketch, *c*, with heat-exchanger, intercooling and reheating, represents a combination having an efficiency of about 30 per cent.

Semi-Closed Cycle

The semi-closed, constant-pressure cycle, developed by Sulzer Bros., is shown in Fig. 7. In this sketch *AH* represents the air heater, *C* the compressor, *HE* the heat-exchanger, *PC* the precooler and *T* the turbines. The compressed air, after being preheated in the heat-exchanger, divides into two parts—one passes to the combustion chamber and is employed for combustion of the fuel oil, whereas the other passes through an air heater which is heated by gases from the combustion chamber. In this way the combustion gases are cooled to about 1200 F before passing to the power turbine where they are expanded down to atmospheric pressure and discharged. That part of the

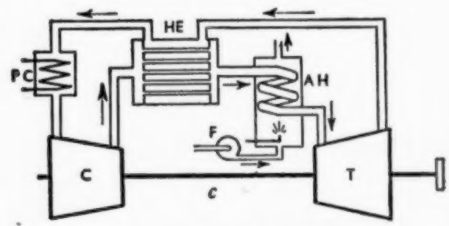


Fig. 8—Closed cycle

compressed air that flows over the air-heater tubes is also raised to about 1200 F. This air does not become polluted with combustion gases and is expanded in a turbine down to a medium pressure, thence to the heat-exchanger. It is finally cooled to a low temperature in the water-cooled precooler and enters the second compressor.

Closed Cycle

The closed cycle, as developed by Escher Wyss, is shown in Fig. 8. Here the reference letters are the same as in Fig. 7 with *F*, the combustion-air fan, added.

In this cycle the compressed air, after passing through the heat-exchanger, is heated indirectly in the air heater. This air does not come in contact with the combustion gases, hence the turbine and heat-exchanger will remain clean and free from corrosion. After the air has left the turbine and passed through the heat-exchanger, it is cooled to a low temperature in a water-cooled precooler before entering the compressors which return it again through the heat-exchanger to the air heater. The closed cycle is thus completely sealed off from the atmosphere and, while the plant is running at steady load, only a small quantity of air has to be supplied to make up for leakage losses.

Nuclear Energy Expert Discusses Its Possibilities

Speaking at the A.S.M.E. Spring Meeting in New Orleans on March 1, Dr. L. B. Borst, chairman of the Nuclear Reactior Project at Brookhaven National Laboratory, Upton, Long Island, predicted that it would be ten to twenty years before atomic energy can compete with coal as a source of industrial power in this country, although it may possibly become economically feasible in Europe somewhat sooner because of fuel shortages abroad.

Notwithstanding this, he ventured the opinion that the generation of power from the atom may be demonstrated at Brookhaven within the next two years. However, since the nuclear reactor of which Dr. Borst will have charge, is planned for research rather than to serve a power plant, the power generated will be a by-product. Contracts are being negotiated for a steam plant to derive its energy from the nuclear pile, and this plant will generate electricity for driving cooling fans and for other apparatus connected with the project. About two and a half billion dollars have been expended so far, mostly through industrial contracts given out by the Commission.

Some Problems to Be Solved

Among current problems awaiting solution is that of operating reactors at sufficiently high temperature for use with the conventional heat engine. Also, according to Dr. Borst, the scientists have long been looking for some trick method of getting electrical energy direct from the chain reaction, but so far their efforts have not met with success.

It is generally acknowledged that generation of electric power, for the foreseeable future, must employ the heat engine. But when using available materials satisfactory from a nuclear standpoint, it has not been possible to operate the reactors at appropriate temperatures. Most common engineering materials, both ferrous and nonferrous, absorb neutrons, hence may not be used inside thermal neutron reactors. Those which do not absorb neutrons lack mechanical strength and undergo corrosion at the necessary temperatures. The few exceptional materials having the desired

qualities are extremely expensive. Studies are now underway toward overcoming these difficulties.

A second problem has to do with the reprocessing at intervals of the nuclear fuel to remove the so-called "ash." At present this must be done behind thick concrete walls, entirely by remote control.

A third problem has to do with fissionable material economy. In the utilization of uranium, only one atom in every 140 is the isotope U 235 which undergoes thermal neutron fission. The other 139 are all U 238 which absorbs neutrons to make plutonium. The principal reactors throughout the country are based on the fission of U 235 and therefore the use of this material is less than one per cent efficient. Thorium is not used; and, although uranium is abundant, high-grade deposits are rare. It was Dr. Borst's opinion that for large-scale power we must eventually learn to use U 238 as well as thorium. Experiments now under way will reveal whether this is possible.

Fission products, or "ash," may become abundant sources of radioactive elements, to be used both in industry and in medicine.

Carbon[14], a radioisotope, is being used in several universities to study carbohydrate, fat and protein metabolisms.

Calling for industrial support of the Atomic Energy Commission's program, Dr. Borst warned that "bottling up the enterprise behind a wall of secrecy might well jeopardize national security rather than aid it."

Large Turbines Building in England

Turbine builders in England appear to be loaded with orders for many large machines, in addition to many of small and medium capacity. According to a listing in *Engineering and Boiler House Review*, the two principal builders of large turbine-generators now have on order ten 60,000-kw, six 50,000-kw and two 45,000-kw machines for installation in British power stations alone, as well as a considerable number for export. The 60,000-kw and 50,000-kw units will be of the three-cylinder type and most of them are being designed for 900 psi, 925 F.

Opportunities for Naval Officers

The U.S. Naval Academy Alumni Association has sponsored a Naval Officers' Placement Service which is available to naval officers, active or retired, as well as those holding commissions in the Naval Reserve, who may be looking for civilian employment. Several hundred good openings are reported, and interested applicants should register with this Service, which is located at 1129 Vermont Avenue N.W., Washington, D.C.

C. E. C. Executives Assume New Duties

J. V. Santry, president of Combustion Engineering Company, New York, announces that John Van Brunt, vice president in charge of engineering for the past twenty-five years, has been appointed vice president and consulting engineer, and that Wilbur H. Armacost, vice president, has been appointed to succeed Mr. Van Brunt as vice president in charge of engineering.

John Van Brunt

Mr. Van Brunt, a graduate of Stevens Institute of Technology of the class of 1897, has long been recognized as an outstanding authority on the design of large high-pressure boilers and fuel burning equipment, and has been responsible for the design of many hundreds of boilers now in service in large power stations throughout the country. He is a fellow of the American Society of Mechanical Engineers and a member of various other engineering societies.

Mr. Armacost, a graduate of Armour Institute of Technology of the class of 1916, has been associated with the Company and its affiliates since 1920. He was elected a vice president of Combustion in 1943.

Wilbur H. Armacost

(Continued from page 31)

generally recommended in order to provide ample protection for the superheater under all operating conditions; although in some cases it might be satisfactory to reduce this differential to 5 lb. The probable settings would be as follows:

Valve on superheater set at 890 psi, closing at 859 psi
First valve on drum set at 890 psi, closing at 868 psi
Second valve on drum set at 910 psi, closing at 878 psi

If only the valve on the superheater were relieving, the pressure at the superheater outlet would be reduced to 859 psi, which is above the normal operating pressure of 838 psi. If, however, the unit were operating at full rating, so as to have the full 35-lb pressure drop through the superheater, the first valve to pop would be the one on the drum set at 900 psi. It would not close until the drum pressure had fallen to 868 psi, which would mean 868 − 35 = 833 psi at the superheater outlet, or 5 lb under the 838 psi normal operating pressure. However, this is the best that can be had under the assumed conditions.

For such a case, a safety valve of smaller per cent blowdown would be desirable. A power control valve or a pilot-actuated valve might be used as both may have as little as 1 per cent blowdown. There is still also another safety valve of the spring-loaded type, the manufacturer of which claims it can be set for 1 per cent blowdown.

NEW CATALOGS AND BULLETINS

Any of these publications will be sent on request

Burners

Coppus Engineering Corp. has released Bulletin 410-5 on Fanmix gas burners. This 16-page bulletin discusses the background leading up to the design of the present burner, gives the classification and application of burners in general, and covers the operating principle and advantages of Fanmix burners. The various types for combination gas and oil burning or for gas only are described and shown in detailed sectional views. Methods of selection of burners with examples are given. This is followed by special designs, applications and safety precautions.

Condensers

Steps in the manufacture of an Allis-Chalmers surface condenser are pertinently portrayed in a new 32-page bulletin, "Let's Watch A Condenser Being Built."

Fabricating, fitting, welding and machining of the condenser shell, forming and machining of the flanges, machining of the water boxes, laying out the tube sheets and steady plates, drilling the tube sheets, tubing the condensers, rolling in and flaring the tubes, assembling the completed parts, and finally shipping the completed unit are included.

Liquid Level Indicators

Yarway Bulletin WG-1822 on remote liquid level indicators, replacing older Bulletin WG-1821, has been made available by Yarnall-Waring Co. This 16-page pamphlet lists the outstanding features of this equipment, describes construction and installation details, and is illustrated with cut-away views, diagrams and installation photographs. Also included is a discussion of the control unit for remote signalling and controlling devices, together with illustrations and control unit application wiring diagrams.

Oil Burners

Schutte & Koerting Co., Philadelphia, has brought out a new three-color, 8-page bulletin on its mechanical fuel oil burners for power boilers. In addition to the descriptive matter there are tables, drawings, charts and data giving dimensions, pressures, capacities and weights, as well as information on pumping systems and air control registers.

Tube Cleaners

The complete new line of "Rotojet" tube cleaners, manufactured by the Elliott Company's Roto Division, 160 Sussex Ave., Newark, N. J., is described and illustrated in a new bulletin just off the press. Also shown are various types and sizes of motors for operation with water, compressed air or steam.

Steam Turbines

"Standardized Steam Turbine Units" is the title of a new 16-page catalog issued by Allis-Chalmers containing essential information and illustrations on six sizes of standardized units, namely, 11,500, 15,000, 20,000, 30,000, 40,000 and 60,000 kw. Included are lists of the features and accessories furnished with these standardized turbines with air-cooled or hydrogen-cooled generators. They are designed in accordance with the preferred A.I.E.E.-A.S.M.E. Standards for steam turbines and generators larger than 10,000 kw.

Water Level Control

Northern Equipment Co. has issued Bulletin No. 475 describing its boiler water-level control as installed at the plant of the Hammermill Paper, Erie, Pa. The Copes "Flowmatic" control, with relay-operated valves, is applied to two 175,000-lb per hr boilers operating at 675 psi.

Water Conditioning

The Liquid Conditioning Corp. announces General Catalog G-1 which is a 60-page bulletin revised to include some recent developments in the design of water treatment and liquid conditioning equipment. Each of the many different types of water conditioning processes are described together with the applications, advantages and limitations of each. Included are tables listing various kinds of gaseous and solid impurities and a chart showing the chemical results produced by various water treatment methods. Numerous diagrams showing the principles of construction and operation of the various processes and equipment are also included.

"Automatic Equipment for Pressure and Level Control" is the title of Catalog 47 issued by Kieley and Mueller, Inc. It is comprised of five separate bulletins covering Kontrol Motor Diaphragm Valves, liquid level controllers, strainers, pressure reducing and regulating valves and other steam plant equipment. The booklet is well illustrated with diagrams, charts of dimensions and weights, performance curves and other views.